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# Small Screw Compressors for Automobile Air Conditioning Systems

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# SMALL SCREW COMPRESSORS FOR AUTOMOBILE AIR-CONDITIONING SYSTEMS

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## ABSTRACT

Following features are required to the compressor for the automobile air-conditioning systems :
(1) larger cooling capacity
(2) less power consumption
(3) smaller both in size and weight
(4) quieter when operated
These requirements will become more severe in the future with the shortage of the fossil fuel. Among the positive displacement type compressors, screw compressors seem to satisfy these demands from the automobile manufacturers with their promissing characteristics with an unloader device.
Mitsubishi Heavy Industries Co. Ltd. has been studying and developed small screw compressor for automobile air conditioner, some of the theoretical and experimental results of which are presented in the paper.
Further advantage of the screw compressor will be obtained with the improvement as follows:
(1) Reducing compressor size and weight by high speed operation.
(2) Increasing system's efficiency by applying the economyzer system.
(3) Developing a rotor profile with smaller internal leakage area.

## INTRODUCTION

Now, reciprocating compressors and axial piston compressors have been widely used for automobile air-conditioning systems. In the last few years, screw compressors and rotary vane compressors have come into the market.
The main reason of their appearance is a change of requirements to the compressor for the automobile air-conditioner.

(1) Power Saving
Since oil shock in 1974, fuel price has been rising year by year, and it is expected that this tendency will continue. Therefore, automobiles and air-conditioning systems are required to have higher efficiency, lighter weight, and power saving device.

(2) Smaller Both in Size and Weight
The 1976's air pollution regulation in Japan and a trend to design larger room space in an automobile result in the limitation of the room space of engine. Smaller compressors are being required.

(3) Quieter Operation
Nowadays, more than 60 percent of automobiles are equipped with air-conditioner in Japan. Users request more deluxe and more comfortable air-conditioning systems operating quietly.

(4) Larger Cooling Capacity
Users request idle-cooling and quick pull-down cooling.

We have been studying the screw compressors from these points of view since 1973 and have developed them.
This paper shows the features and the performance of our small screw compressors.

## CONFIGURATION-SMALL SIZE AND LIGHT WEIGHT

Fig. 1 shows a cross section of the screw compressor for cars (type MSN653), and Fig. 2 shows for a buses (type MSN8513FR).
Their specifications are tabulated in Table 1.
The compressor type MSN653 has a pair of rotors, casings, oil separator, oil injection valve, relief valves, check valves, gas temperature sensor and a magnetic clutch.
The features are as follows :

(1) Small Size
This compressor uses 47ømm male rotor with unsymmetric profile, and is driven by 6-tooth female rotor, namely it has 6-compressions per revolution. It has a compact oil separator and oil injection valve actuated with pressure difference.

(2) Light Weight
A front casing and rotor casing are made of aluminium die-casing.

The compressor type MSN8513FR has a pair of rotors, casings, an unloader device, an oil injection valve and a magnetic clutch.
The features are as follows :

(1) Small Size and Light Weight
This compressor uses 85ømm male rotor with unsymmetric profile, and a detached high

quality oil separator. It is directly mounted on a diesel engine body. Its small size and light weight are excellent merit.

(2) Unloader Equipment
It has an unloader device, which gives the advantage of capacity control and power saving.

LARGE COOLING CAPACITY

The comparison of cooling capacity and efficiency between the screw compressor and other type compressors is shown in Table 2.
In Table 3, the evaporator outlet air temperature is tabulated in case of the screw compressor and the 2-cylinder reciprocating compressor.
Table 4 shows the compressor weight per cooling capacity.

(1) Cooling capacity and the volumetric efficiency of the screw compressor at high speed operation, are the largest among other type of compressors listed in Table 2.
The reason is that it has no discharge and suction valve and no top clearance volume, that are inevitable in case of reciprocating compressors. At high speed operation, the screw compressor has better performance than 4-sliding vane rotary compressor because of no discharge valve and no sliding vane.
(2) The internal leakage of the screw compressor, from which the reciprocating compressor is free, has influence on the performance at low speed operation. In order to solve this problem, our compressor uses polyglycol oil, the viscosity of which is 92 cSt at 100°C, and viscosity index, 240. To keep oil from being held up in heat exchangers and pipings, a high efficiency oil separator is necessary.
From Table 3, it is seen that cooling capacity at low speed operation is nearly equal to that of the 2-cylinder reciprocating compressor.

QUIET OPERATION

The sound pressure level of the various compressors at a driver's ear point in the same compact car is shown in Fig. 3, and the sound power level measured by bench tests, in Fig. 4. From these figures, it is seen that the sound level of the screw compressor is lower than any of the 2-cylinder reciprocating and the 6-cylinder axial piston compressors, and is almost the same as the 4-sliding vane rotary compressor. Data at more than 3000 rpm are not shown for the reason that the back ground noise level is larger than that at the compressor running. The reason of quiet operation is as follows :

(1) This screw compressor is driven by 6-tooth female rotor, which results in the small fluctuation of the torque.
(2) Discharge gas pulsation is about 0.1 $kg/cm^2$ (peak to peak) because of damping by the oil separator housing volume. This is nearly one-third of the pulsation in the 2-cylinder reciprocating compressor.
(3) Suction gas pulsation is negligibly small being damped by the suction chamber volume.

POWER SAVING-UNLOADER EFFECT

The compressor type MSN8513FR has the unloader device, which modulates cooling capacity to match the passengers' number and improves a fuel consumption rate. Fig. 5 shows a working principle of the unloader device. This device consists of sensor, a magnetic 3-way valve, and an unloader piston. The system low pressure or the evaporator outlet air temperature is transformed into the electric signal. In case of full load operation, the magnetic valve makes a discharge gas work on an unloader piston, which closes gas bypass holes through a rotor casing wall. For the part load operation, the discharge gas is ventilated to the suction side through the magnetic valve, and the pressure difference between both ends of the piston becomes almost zero. A spring force moves the piston, and the gas in the cylinder is bypassed through the hole, so that the effective cylinder volume is decreased.
Fig. 6 depicts the unloader performance curve. It is seen that a reduction rate of the cooling capacity depends mainly on the speed, and a power consumption rate mainly on an operating pressure ratio.
The reason is as follows :

(1) At a constant operating pressure ratio, a gas bypass flow rate is nearly constant and independent of the speed. Accordingly, the capacity reduction rate decreases with increasing the compressor speed.
(2) An adiabatic efficiency of the screw compressor depends on a built-in volume ratio, i.e. a built in pressure ratio and an operating pressure ratio. The efficiency decreases with the increase of the difference between the operating and design pressure ratio.

Fig. 7 shows the calculated results of cooling capacity and power in this air-conditioning system. It is seen that at 100 km/h, a cooling capacity reduction is almost 10%, and power saving, almost 1 PS. The reduction rate of power increases with decreasing outdoor temperature and compressor speed. In Table 5, the ratio of fuel consumption rate, measured by a car at 100 km/h on a dynamometer, is tabulated. It is shown that the ratio of fuel consumption rate at compressor full load operation to that of the air-conditioner off operation is 1.12, and the ratio at compressor part load to that of no load is 1.09. Therefore the ratio at part load to full load is 0.97.

Namely, the unloader device decreases the fuel consumption rate by 3% as compared with no unloader device. The advantage of the unloader device will become more distingushed if the fuel saving is considered throughout the year.

COMPRESSOR COMPUTER SIMULATION

As described previously, the efficiency of the screw compressor and the rotary compressor depends mainly on the internal gas leakage, especially at low speed operation.
To analyze the performance of the screw compressor, the internal leakage cannot be ignored. As far as

this type of compressor is concerned, theoretical approach to considering the internal leakage is important and have not been studied too much. The examples of our theoretical approach are as follows.

(1) Modeling & Basic Equations

Fig. 8 shows a schematic diagram of the leakage flow paths in this screw compressor. The following equation is derived from the mass balance and energy balance, as the cylinder volume is given.

$$\frac{dT}{d\varphi}=\frac{1}{(\partial u/\partial T)_v}\cdot\frac{1}{G}\Big[\frac{dQ_w}{d\varphi}-\Big\{\Big(\frac{\partial u}{\partial v}\Big)_T+Ap\Big\}\Big\{\frac{dV}{d\varphi}-v\sum_i\frac{dG_{in,i}}{d\varphi}$$

$$+v\sum_j\frac{dG_{out,j}}{d\varphi}\Big\}+\sum_i h_{in,i}\frac{dG_{in,i}}{d\varphi}-h\sum_i\frac{dG_{in,i}}{d\varphi}\Big]$$

where

| | |
|---|---|
| T : absolute temperature | Qw : heat flow into the cylinder |
| P : absolute pressure | Gin, Gout : weight of gas flow into, out of the cylinder |
| U : internal energy | i,j: suffix of the internal leakage flow path |
| h : enthalpy | A : mechanical equivalent of heat |
| v : specifie volume | |
| G : weight | |
| V : cylinder volume | |
| $\varphi$ : rotating angle of the rotor | |

The leakage is treated as nozzle flow, and the flow restrictor coefficients are determined from experiments. The leakage gas enthalpy is assumed constant before and after leakage. It is easy to calculate, ($dGin/d\varphi$), ($dGout/d\varphi$), ($dV/d\varphi$), and ($dQw/d\varphi$), so that the equation above can be integrated numerically by a digital computer.

(2) Flow Chart

Fig. 9 shows the flow chart of this computer program, features of which are as follows.

(a) The effects of internal leakage are considered.
(b) Thermodynamic properties of a refrigerant are treated as those of a real gas.

(3) Calculation Results

Calculated results considering internal leakage and those assuming ideal isentropic process without leakage is shown in Table 6 and Fig. 10. Comparing both results, it is seen that considering leakage is important. Fig.10 shows the pressure and temperature change of the gas in the cylinder with the rotor rotation. In comparison to the isentropic process without leakage, they increase more quickly in case of considering leakage. The effects of clearance at the leakage path is an important information in designing and manufacturing compressors. Fig. 11 shows the examples of those effects on a volumetric efficiency. A degree of effects of each leakage path is not same but different. Useful information obtained from this calculation have been reflected to the design and manufacture. This computer simulation should be extended to describe more total behaviour of the compressor.

## FUTURE DEVELOPMENT

As described here, the screw compressor has much merit for automobile use. The advantages of the screw compressor over any other type will become more clear with the following improvement in the future.

(1) Reducing compressor size and weight by high speed operation.
(2) Increasing system's efficiency by applying an economizer system.
(3) Developing a rotor profile with smaller internal leakage area.

Table 1. Specifications of Screw Compressors

| COMPRESSOR | MSN653 | MSN8513FR |
|---|---|---|
| PHYSICAL DISPLACEMENT | 106 cc/rev. | 578 cc/rev. |
| EQUIVALENT DISPLACEMENT | 138 cc/rev. | 750 cc/rev. |
| WEIGHT (WITH CLUTCH) | 8.3 KG | 39.6 KG |
| ROTOR DIAMETER (MALE ROTOR) | 47ø mm | 85ø mm |
| LOBE COMBINATION | 4(MALE) + 6(FEMALE) | 4(MALE) + 6(FEMALE) |
| ROTOR PROFILE | UNSYMMETRIC | UNSYMMETRIC |
| WRAP ANGLE | 250° (MALE) | 250° (MALE) |
| BUILT-IN VOLUME RATIO | 5 | 3.5 |
| DRIVE | FEMALE ROTOR DRIVE | FEMALE ROTOR DRIVE |
| DIRECTION OF ROTATION | CLOCKWISE DIRECTION | CLOCKWISE DIRECTION |
| OIL INJECTION | BY PRESSURE DIFFERENCE | BY PRESSURE DIFFERENCE |
| OIL SEPARATOR | INCLUDED | SEPARATED |
| REFRIGERANT | R - 12 | R - 12 |
| OIL BRAND | RS - 680 | RS - 680 |

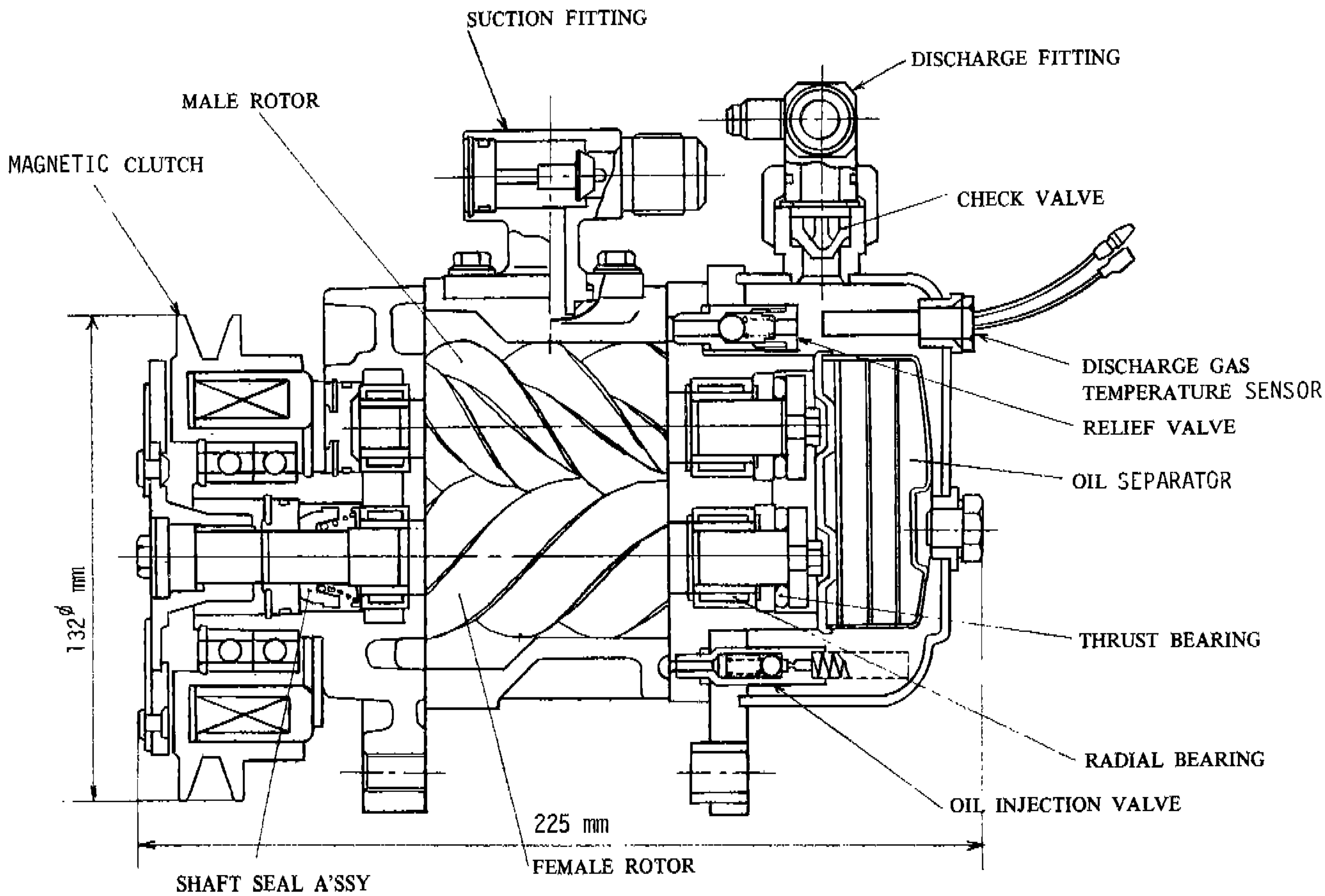



Figure 1. Screw Compressor (MSN653)

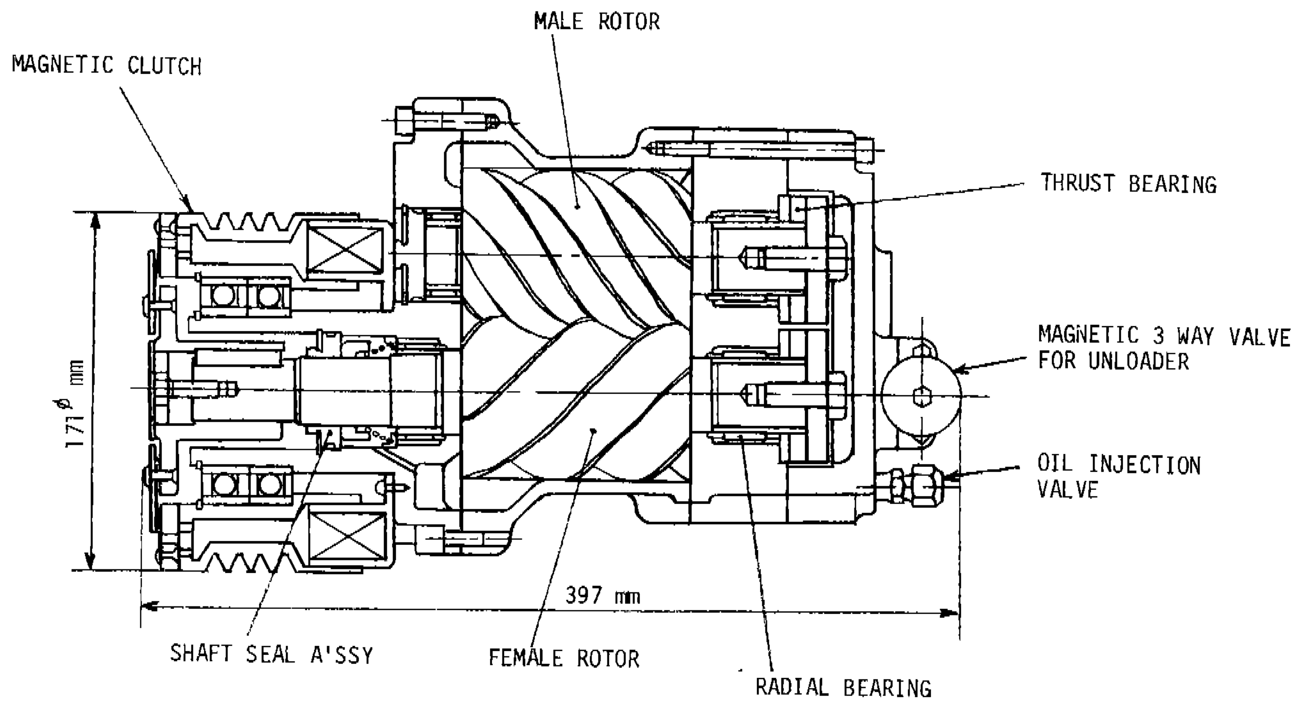



Figure 2. Screw Compressor (MSN8513FR)

Table 2. Compressor Comparison

| DESCRIPTION \ COMPRESSOR TYPES | | SCREW (MSN653) | ROTARY | 6-CYLINDER AXIAL PISTON | 2-CYLINDER RECIPRO. |
|---|---|---|---|---|---|
| PHYSICAL DISPLACEMENT | cc/rev. | 106 | 115 | 134 | 107 |
| WEIGHT KG | TOTAL | 8.3 | 9.4 | 8.4 | 8.7 |
| | WITHOUT CLUTCH | 6.2 | 6.8 | 5.8 | 5.5 |
| 800 RPM RATING | CAPACITY Kcal/H | 1,540 | 1,650 | 1,470 | 1,550 |
| DST/SST - 72/3°C<br>Ts = 14°C<br>S C = 5 degC | VOLUMETRIC EFFICIENCY % | 73 | 72 | 55 | 72.5 |
| A T = 100°C | POWER KW | 1.17 | 1.39 | 1.06 | 0.0843 |
| | ADIAVATIC EFFICIENCY % | 55 | 50 | 58 | 76 |
| 1,000 RPM RATING | CAPACITY Kcal/H | 2,280 | 2,370 | 2,360 | 2,000 |
| DST/SST = 60/0°C<br>Ts = 18°C | VOLUMETRIC EFFICIENCY % | 83.5 | 80 | 68.5 | 73 |
| S C = 5 degC | POWER KW | 1.20 | 1.54 | 1.16 | 0.874 |
| A T = 20°C | ADIAVATIC EFFICIENCY % | 63.5 | 51.5 | 68 | 76 |
| 2,500 RPM RATING | CAPACITY Kcal/H | 3,130 | 3,024 | 2,950 | 1,900 |
| DST/SST=55/-17°C<br>Ts = 18°C | VOLUMETRIC EFFICIENCY % | 80 | 71 | 60 | 48 |
| S C = 5 degC | POWER KW | 2.40 | 2.87 | 2.17 | 1.32 |
| A T = 20°C | ADIAVATIC EFFICIENCY % | 57 | 46 | 59 | 62 |
| MAXIMUM SPEED | RPM | 8,000 | 6,500 | 6,000 | 6,000 |

NOTE: DST ... Discharge saturation temp.
Ts ... Suction gas temp.
A T ... Compressor ambient temp.
SST ... Suction saturation temp.
S C ... Degree of subcooling

Table 3. Cooling Capacity

| TEMPERATURE OF EVAPORATOR OUTLET AIR °C | SCREW | 2 CYLINDER RECIPRO. |
|---|---|---|
| SOAK OUT | 40°C DB 80% R.H | 40°C DB 80% R.H |
| 40 Km/H RUN at 5 MIN.PASSED | 17.0 | 17.0 |
| at 10 MIN.PASSED | 12.0 | 12.0 |
| at 30 MIN.PASSED | 6.9 | 7.0 |
| 100 Km/H RUN at 60 MIN.PASSED | 2.5 | 7.0 |
| IDLING at 90 MIN.PASSED | 13.5 | 14.0 |

NOTE: (1) Test car is equipped with 2.0ℓ - 4 cylinder gasoline engine.
(2) Load ; 40°C DB, 80% R.H
(3) Test was run at a dynamometer
(4) Compressor size and performance are described in Table 2.

Table 4. Compressor Weight per Cooling Capacity at 2500 RPM Rating

Kg/Kcal/H

| SCREW | ROTARY | 6 CYLINDER AXIAL PISTON | 2 CYLINDER RECIPRO. |
|---|---|---|---|
| $2.65\times10^{-3}$ | $3.10\times10^{-3}$ | $2.85\times10^{-3}$ | $4.58\times10^{-3}$ |

NOTE: Compressor size and performance are described in Table 2.

Table 5 Ratio of Fuel Consumption Rate

| | AT FULL LOAD / AT AIR-CON. OFF | AT UNLOAD / AT AIR-CON. OFF | AT UNLOAD / AT FULL LOAD |
|---|---|---|---|
| RATIO | 1.12 | 1.09 | 0.97 |

NOTES: The fuel consumption rate was measured by a car running at 100 km/H on a dynamometer.
Pulley ratio is 1.0.
Outdoor temperatures is 20°C.

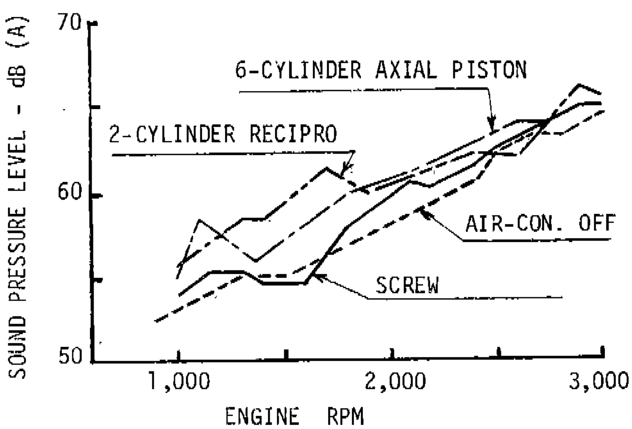



Fig. 3 Sound Pressure Level at a Driver's Ear Point in a Car (2.0ℓ - 4 cylinder gasoline engine)

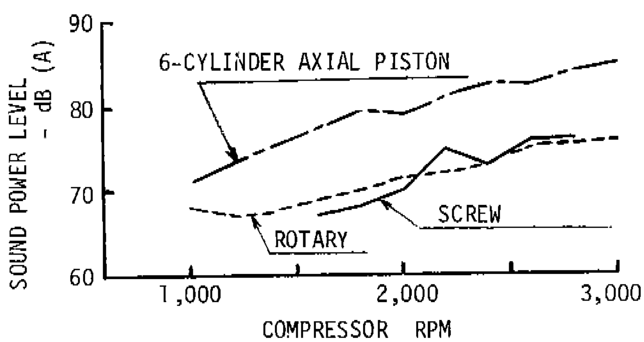



Fig. 4 Compressor Sound Power Level

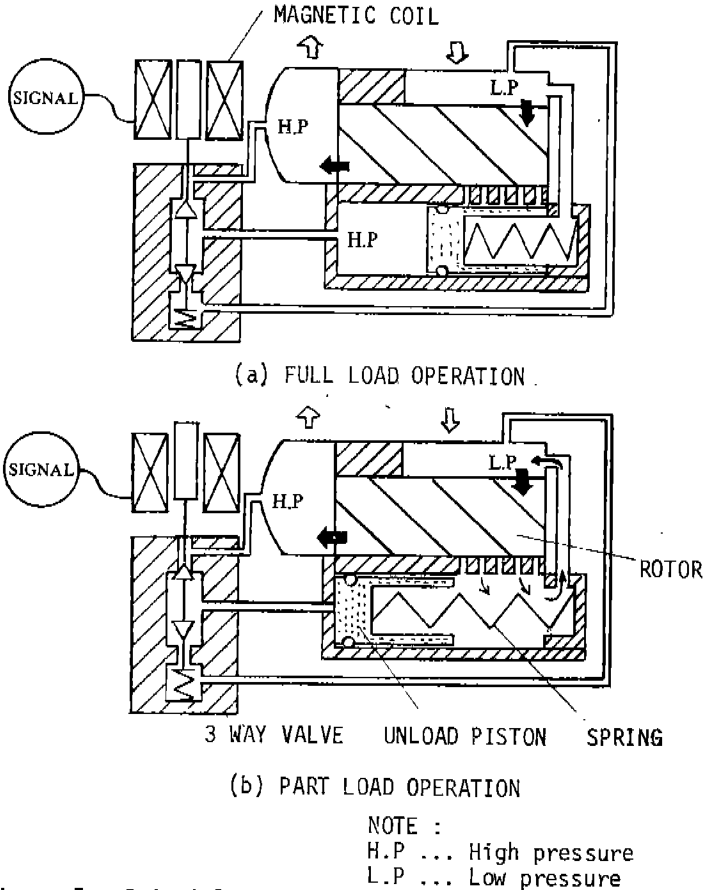



Figure 5 Principle of Unloader Mechanism

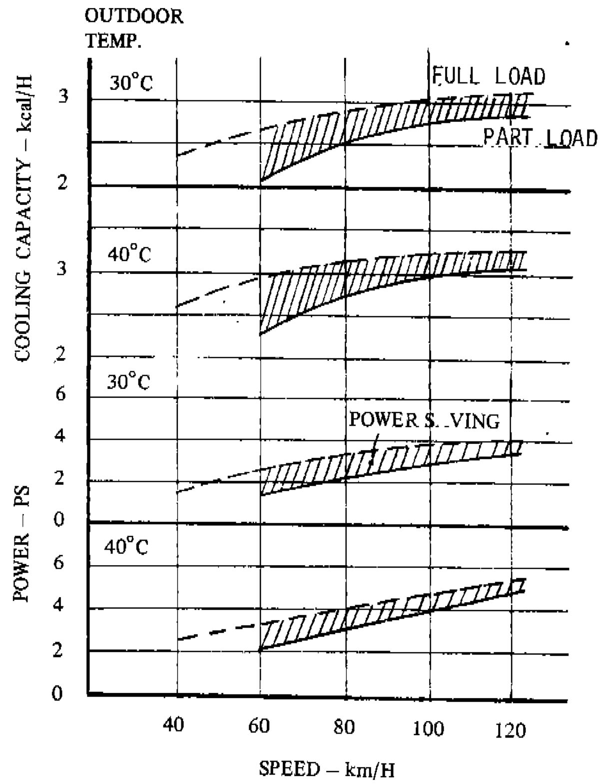



**Figure 7 Calculation Result of Unloader** Effect

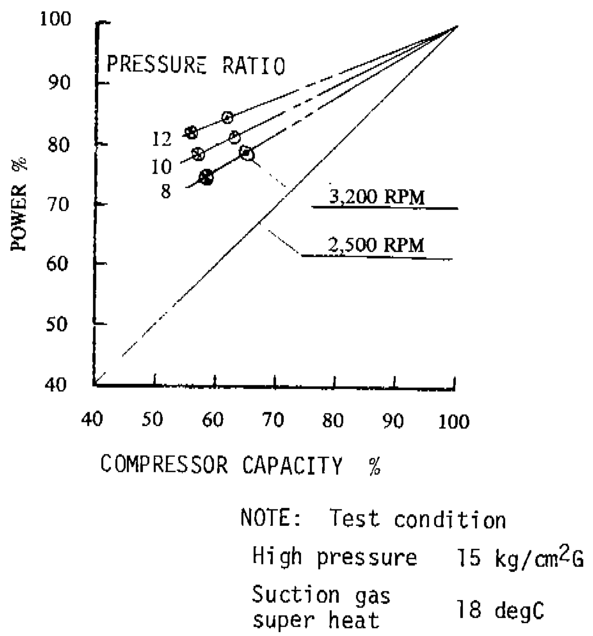
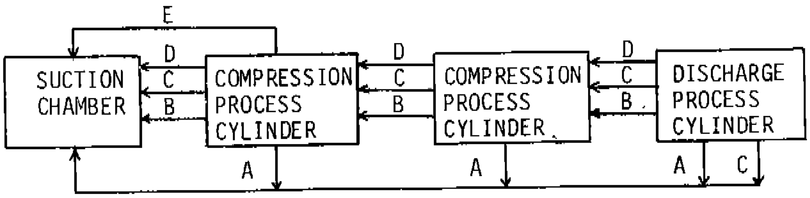



Figure 6 Unload Characteristics




NOTES:
A : Rotor meshing portion
B : Rotor tip portion
C : Rotor discharge side portion
D : Blow hole portion
E : Rotor suction side portion

Fig. 8 Schematic Diagram of the Compressor Internal Leakage Paths

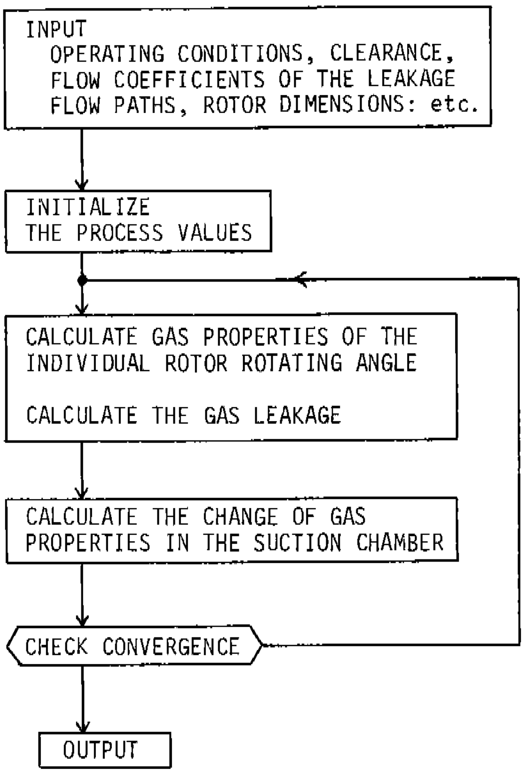



Fig. 9 The Flow Chart of the Digital Computer Program

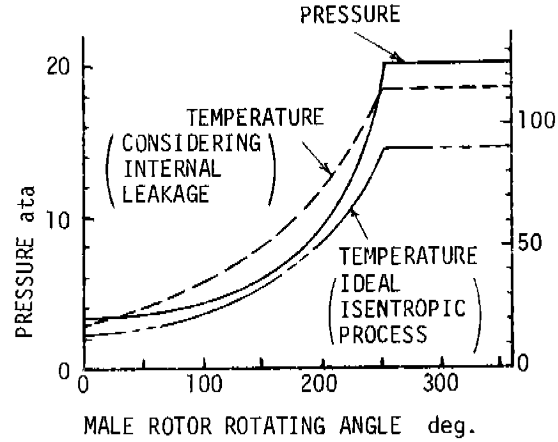



Fig. 10 Calculation Results on Pressure and Temperature

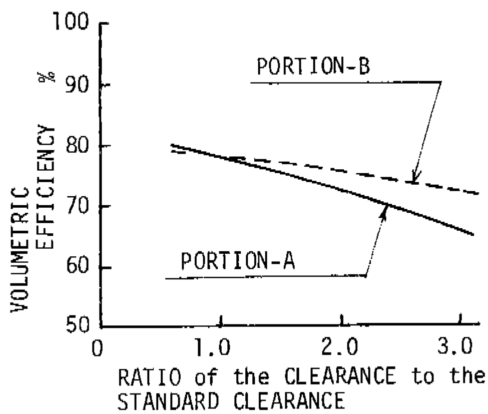



Fig. 11 Calculation Results on the Effects of Clearance of Leakage Path

Table 6 Calculation Results on the Compressor Performance

| ITEMS | | CONSIDERING THE INTERNAL LEAKAGE | ASSUMING THE ISENTROPIC PROCESS |
|---|---|---|---|
| VOLUMETRIC EFFICIENCY | % | 78 | 100 |
| COOLING CAPACITY | Kcal/H | 1660 | 2100 |
| POWER | Kw | 1.20 | 0.90 |
| ADIAVATIC EFFICIENCY | % | 59 | 100 |

NOTICE: (1) At 800 RPM Rating with the normal manufacturing clearances
(2) Thermal deformation and heat exchange between the compressor and surroundings are neglected.